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Title of Invention:

MULTI-STAGE ELECTRIC PUMP UNIT

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To All Whom It May Concern:
The following is a specification
of the aforesaid Invention:

MULTI-STAGE ELECTRIC PUMP UNIT

Multi-stage electric pump units are hydraulic machines that consist of multi-stage centrifugal pumps, the electric motor of which is directly coupled to the pump part.

The main hydraulic elements are the impeller and the diffuser. The impeller is the key element in every pump (it converts the rotation speed into pressure), although the body, or diffuser, is highly important, given that it redirects the fluid from the impeller outlet to the inlet of the next impeller, thereby recovering the static pressure. When the desired head is very high, various impellers are positioned in a series, thus the designation, clean water multi-stage units.

What is demanded of an electric pump is that it provide maximum efficiency, and this will be achieved if the impeller and diffuser provide maximum efficiency.

The efficiency rating of modern electric pumps is usually between 75% and 85% of the total work input into the electric pump.

The objective of this invention is to exceed that efficiency.

The inventor has considered the following to be fundamental: that the pump reach stability, that cavitation and turbulence be prevented, that the curvature and non-curvature zones of the impeller and diffuser vanes be such that the flow is guided perfectly and that this flow correctly follow the direction of the vanes.

The applicant has developed a multi-stage electric pump unit, of those that consist of a multi-stage centrifugal pump and an electric motor directly coupled to the pump, where each stage of the pump consists of an impeller and a diffuser that each have channels with vaned and vaneless zones. These zones are delimited between a shroud-surface and a hub-surface, where the β angle is the angle of the tangent at each point, which is characteristic because, given that the Y-axis of the electric pump is the radial co-ordinate and X is the axial co-ordinate, for flow rates Q between 2500 and 8000 l/min., the points of the surface of the impeller and of the diffuser comply with the following sixth-degree polynomial equation: $Y = f(x) = Ax^6 + Bx^5 + Cx^4 + Dx^3 + Ex^2 + Fx + G$, and where on the diffuser:

a) on the hub, the vaneless zone; A=B=C=D=E=0; F=0.6605; G=20.45;

b) on the shroud, the vaneless zone; A=B=C=D=E=0; F=0.7225; G=55.648

c) on the hub, the vaned zone; A=-9E-09; B=7E-06; C=-0.0019; D=0.3064; E=-26.923; F=1256.3; G=-24283

d) on the shroud, the vaned zone; A=1E-10; B=-9E-08; C=2E-05; D=-0.0033; E=0.2349; F=-7.616; G=174.28

e) on the hub, the vaneless zone; A=0; B=0; C= -1E-05; D=0.0073; E=-1.7542; F=186.27; G=-7311.6;

f) on the shroud, the vaneless zone; A=0; B=0; C=0; D=0.0053; E=-2.6745; F=446.37; -G=24717;

g) on the hub, β; A=0; B=0; C=1E-06; D=-0.0002; E=0.0203; F= -1.0819; G=156.82;

h) on the shroud, β; A=0; B=0; C=3E-07; D= -1E-04; E=0.0101; F=-0.7587; G=175.

For different flow rates, the constants of the polynomial are modified for the impeller.

With these geometric characteristics of the electric pump components, the efficiencies obtained with the electric pumps are substantially higher than currently known efficiencies, thereby reaching increases of up to 14%, which is surprising in these kinds of machines.

In conventional technology, both the impeller and the diffuser are made of a casting, thereby preventing the precision of the geometric shapes/curves, which have characteristics that are so difficult and special to both bodies.

In the invention, and once the aforementioned polynomial equation is developed, a machining centre can be programmed precisely, thereby achieving the fewest deviations by the actual geometric shapes with respect to the theoretical shapes, and as a result, very small deviations by the actual efficiencies with respect to the theoretical efficiencies.

In order to understand the object of this invention better, explanatory figures have been attached.

Figure 1 is a representation of a vortex-whirlpool at the intake (a) of the fluid (f).

Figure 2 is a representation of the limit layer of the fluid, which is the channel wall of the impeller, for example.

Figure 3 is a representation of fluid inlet onto the impeller vanes.

Figure 4 is a representation of the connection between an impeller and a vaneless diffuser at the vaneless part of a diffuser.

Figure 5 is a cross-sectional and co-ordinated view of the geometry and flow of an impeller.

Figure 6 is a cross-sectional and co-ordinated view of the geometry and flow of a diffuser.

Figure 7 is a diagram of β and the M% for the shroud (T) and hub (C).

The nomenclature used is the following:

- $A \in [2.3, 5]$ This means that A belongs to a closed interval (the end values of the interval are included), which end values are 2.3 and 5. In other words, A can take any value encompassed between 2.3 and 5, both inclusive.

- Q means the flow rate, the units of which are in l/min.

- The plane (Z, R) is the plane defined in figures 5 and 6: it is graphed by R ordinates (radial co-ordinate) and Z abscissas (axial co-ordinate) (R and Z expressed in mm). The Z-axis is offset on the graph with respect to the Y-axis of electric pump revolutions.

- The shroud and hub are the curves shown in the figure, for both the impeller and the body or diffuser. All of these curves show axisymmetry (axial symmetry).

- β angle (°) (figures 3 and 7).

β on a point of the vane is the angle of the tangent to the vane at the said point (tangential direction).

The graph (β, M%) provides values of β for each value of M% (percentage of the distance travelled over the vane, hereby understanding that 0% is the start of the vane and 100% is the end of the same).

- Impeller and/or diffuser channel.

This corresponds to the volume of fluid enclosed between two consecutive vanes, the shroud and the hub of the impeller and/or diffuser.

- "Double curvature"

This is the magnitude of the tangential co-ordinate from the edge of the vane entry until the edge of the vane exit.

Given a distribution of beta, the "double curvature" values are set at the entry (at both the shroud and the hub) for the complete definition of the impeller channel (likewise for the diffuser).

The constituent elements of the impeller and diffuser are the following:

(T) Shroud: This is the outer-most flow surface of a pump flow path.

Vane: A surface (with a certain distribution of thickness) with an aerodynamic shape that extends from the hub of an impeller or stator and is responsible for the course of the flow, and as regards rotating vanes, for transferring the force to the flow.

(C) Hub: This is the innermost flow surface of a pump flow path.

The walls of the impeller and/or diffuser channel that are not vanes are the shroud (T) and the hub (C).

Once the flow enters the impeller channel, the limit layers begin to develop for the entire vane on the shroud and hub surfaces. The heart of the flow, at least initially, tends to follow the vane channel reasonably well, thereby assuming a uniform input flow. If any distortion occurs at the input flow, then the heart of the flow will be characterised by the presence of vortices from the start, which must be avoided.

Vortices (see figure 1) are an example of potential in which the current lines rotate around a central point (i.e., a low-pressure area, a tornado), wherefore in reality the speed at the centre is null, and therefore pressure is maximum.

The points of contact with the walls of the channel (v = 0) slow down the fluid particles that go close to the walls. A layer called a limit layer is thus formed, within which the speeds of the fluid particles are less than in the undisturbed zone.

$$U_{\text{LIMIT LAYER}} < U_{\text{ALL OTHER PARTICLES}}$$

Where U is the total speed of the particles.

δ is defined as the height of the limit layer (CP) (see figure 3), and it is the geometric site of the points with a speed $\approx$ 99% of the speed of the outer points of the limit layer.

To the extent that the flow advances in the impeller channel, both the heart of the flow (CF) and the surface of the limit layer experience complex force-fields, and they develop more complicated flow shapes. The limit layers initially follow the free-flow direction. However, to the extent that the fluid advances in the impeller channel, both the limit layers and the heart of the flow are influenced by a force field through the channel. It is for this reason that a secondary flow develops around the heart of the flow, which is near the suction surface of the shroud, and a flow with an elevated moment develops in the pressure area of the hub.

In general, a secondary flow is considered a flow in which viscous effects are predominant. Conversely, a primary flow is considered a flow in which the kinetic effects are predominant.

As a general characteristic, it can be affirmed that both the primary flow and the secondary flow comply with conservation equations (equations that, together with the contour conditions, govern the behaviour of the fluid flow as it transfers through the duct, whether rotating or fixed).

The basic fluid dynamics characteristics of a pump stage can be introduced by examining in detail each one of the components and their dominant velocity triangles, using the following principles:

1) The first principle is the mass conservation principle.

2) Newton's equation of movement in a system of angular co-ordinates.

3) Euler's equation of turbomachinery, which simply states that the increase of enthalpy is equal to the change in angular momentum.

With these principles, the velocity triangles at various positions of the flow field can be examined.

The first important point is the inlet (1) between the vanes (2) of the impeller. The main parameters are shown in figure 2.

It is hereby recognised that there is a large variety of possibilities for inlet flow states for a same flow that is pulled towards the eye of the impeller with a meridian speed of C_m.

The flow is found with the pump wheel moving at a perimeter speed of $U = 2 \cdot \pi \cdot r \cdot N$. The velocity triangle is written in accordance with the following basic principle:

$$\text{Relative speed} + \text{Impeller speed} = \text{Absolute Speed}$$

For an inlet flow to the impeller where $C_\theta = 0$ (a perfect inlet of the flow to the impeller), the relative flow angle is defined by the meridian speed at the inlet (controlled by the mass flow and the density variations) and the local speed of the impeller (U). The resulting relative flow angle is β.

If β is equal to the vane angle, then the flow correctly follows the vane direction. However, under different operational conditions, the angle can be greater or less than the vane angle, and as a result thereof, the flow will experience a certain level of incidence. The inlet flow state can also be substantially modified in the event that there is any pre-rotation (rotating water before it enters the wheel), either in the same rotation direction as or contrary to the motor shaft.

This ideal inlet to the impeller can be achieved with an appropriate intake design.

By varying both the mass flow and the flow area, the relative speed W_{1t} (relative speed of the impeller at the inlet (1) to the shroud) has a minimum. To the extent that the speed increases for the inlet diameter or as the flow area increases, U increases. If the flow area is reduced, meaning the mass flow per unit of area, then C_m increases.

The meridian speed component, C_m, is governed by the mass conservation equation, and the absolute flow angle is the result of the relative flow angle and the impeller speed. In practice, the flow angle at the output does not exactly follow the vane, rather it differs by a certain amount. This difference, in terms of tangential speed, is known as the slip speed, from which the slip factor is derived.

For both radial and curved vanes, the resulting absolute flow angle (α) is very large: typically α = 50° to 80° (radial reference) or α' = 10° to 40° (tangential reference). It is most common for this angle to be α' = 15° to 25° and α = 65° to 75°. The velocity triangle at the output is important not only for determining the work level or pressure increase, but also for understanding the variation in pressure with changes in the mass flow.

Once the operating conditions are specified and the impeller efficiency level is known, then the flow can be calculated from the input to the output, in addition to being able to make a good estimate of the slip factor. The resulting flow field is characterised by having a considerable quantity of kinetic energy with a certain rotation at the output.

The efficiency, as previously described, relates the isoentropic work (*isoentropic work*: the least work that could be used to reached the desired head) to the real work. This is a strict definition of thermodynamics.

In order to be able to define the velocity triangles, it is necessary to know the efficiency value of the impeller, and obtaining this is not simple.

The next step to be considered is the discharge from a vaneless diffuser, or at any point within a vaneless diffuser.

It is possible to convert kinetic energy into an increase of the static pressure by changing the radius of the path described for the median flow, in accordance with the law of conservation of angular momentum.

Angular momentum is related to the angular momentum at the output of the impeller in accordance with the law of conservation of angular momentum. In fact, part of this angular momentum is destroyed to the extent that the flow transfers through a vaneless diffuser. Therefore, more precise calculations must be made in order to be able to estimate the level of this degradation. It can be said that approximately 5% to 15% can be lost due angular momentum.

The meridian component of the total speed (C_m) can be precisely calculated based on the mass conservation equation (once the density variation is known). Conversely, the flow angle is determined based on a combination of these two relationships. The flow angle through a vaneless diffuser depends on the width of the channel.

The variation of the area that the flow must follow in the diffuser is used as a second measure of the conversion of kinetic energy into an increase of the pressure energy.

As the connection point between different components, diffusers also play an important role in regulating the flow levels and in controlling the radial thrusts.

Vaneless diffusers (3) (see figure 4) essentially consist of two parallel walls (4) forming an open, annular and radial channel from the impeller output (5) up to some limit with a greater radius. Sometimes it is followed by a volute collector and others by a diffuser body. However, some diffusers have no "pinch," and in other cases a vaneless diffuser is used where the area increases, which is used as part of an input system into a channel diffuser configuration.

The existence of a vaneless diffuser between the impeller and the diffuser body (vaned) helps to prevent coupling of the vibration modes between the

vanes of the diffuser body and the vanes of the diffuser, thereby preventing the edge of the vane of the diffuser body at the input from becoming loaded and failing due to fatigue.

An impeller that is followed by a continuous diffuser body of vanes that begins immediately after the discharge from the impeller and extends to the start of the next stage can also be used.

Considering all of these conditioning factors, the applicant has been performing a considerable number of actual experiments and, computed using the Computational Fluid Dynamics (CFD) technique, has been determining point by point the optimum geometry at each point of the fluid path in a pump in order to achieve optimum efficiency.

With this geometry represented on co-ordinate axes where the Y-axis of the pump is the ordinate reference ($Y \equiv R$) and $X \equiv Z$ is the radial co-ordinate on the Y-abscissas, and applying the mathematical technique to this geometry, he has developed a simple equation, but one that complies with the experimental results obtained and that has an error margin of ± 3.5%.

This equation is the polynomial $Y = Ax^6 + Bx^5 + Cx^4 + Dx^3 + Ex^2 + Fx + G$, where the constants A, B, C, D, E, F and G have to be found for each zone of the pump where the flow is going to travel.

Figure 5 represents the flow scheme and geometry in an impeller that consists of a vaneless zone (6) where the flow coming from the intake enters, and a vaned zone (7) where the flow exits towards the diffuser. The two end surfaces of the shroud (T) and hub (C) can be seen.

Figure 6 represents the flow schematic and geometry in a body or diffuser that, in this case, consists of an initial, vaneless zone (8) where the fluid enters from the impeller output, a vaned zone (9), and a second, vaneless zone (10), which redirects the flow towards the impeller of the next stage.

The flow rates, Q, that have been worked with are in a range from 2500 to 8000 litres/minute.

In the trials performed, the applicant considered that the essential element in the electric pump for reaching optimum efficiency was the body-diffuser, wherefore it remained unchanged, while it was advisable to modify the impeller as the range of the flow rate (Q) was modified. Therefore, flow rate values (Q_1) that are approximately between 2500 and 6000 litres/minute were given to the stated polynomial for a first kind impeller and flow rate values (Q_2) of approximately 4500 to 8000 litres/minute were given for a second kind of impeller.

The equations are the following:

$$\mathbf{Y = f(x) = Ax^6 + Bx^5 + Cx^4 + Dx^3 + Ex^2 + Fx + G}$$

Identification	**A**	**B**	**C**	**D**	**E**	**F**	**G**	**Interval**
Q_1/I/NA/C	0	0	0	6E-05	0.0014	-0.0146	27.511	[1, 40.5]
Q_1/I/NA/T	0	0	0	0	0	0	64.5	[1, 9]
Q_1/I/A/C	0	0	5E-06	-0.0014	0.1535	-6.3821	121.24	[40.5, 83]
Q_1/I/A/T	-4E-08	8E-06	-0.0006	0.0247	-04771	4.3023	50.015	[9, 54.09]
Q_1/I/β/C	0	3E-09	-9E-07	0.0001	-0.0042	-0.0915	34.402	[0, 100]
Q_1/I/β/T	0	1E-09	-5E-07	6E-05	-0.0044	0.1822	22.2	[0, 100]
Q_1 y Q_2/D/NA/C	0	0	0	0	0	0.6605	20.45	[83, 88]
Q_1 y Q_2/D/NA/T	0	0	0	0	0	0.7225	55.648	[54.09, 60]
Q_1 y Q_2/D/A/C	-9E-09	7E-06	-0.0019	0.3064	-26.923	1256.3	-24283	[88, 154]
Q_1 y Q_2/D/A/T	1E-10	-9E-08	2E-05	-0.0033	0.2349	-7.616	174.28	[60, 163.5]
Q_1 y Q_2/D/NA/C	0	0	-1E-05	0.0073	-1.7542	186.27	-7311.6	[154, 174]
Q_1 y Q_2/D/NA/T	0	0	0	0.0053	-2.6745	446.37	-24717	[163.5, 174]
Q_1 y Q_2/D/β/C	0	0	1E-06	-0.0002	0.0203	-1.0819	156.82	[0, 100]
Q_1 y Q_2/D/β/T	0	0	3E-07	-1E-04	0.0101	-0.7587	175	[0, 100]
Q_2/I/NA/C	0	0	0	5E-05	0.0013	-0.0139	27.511	[1, 41]
Q_2/I/NA/T	0	0	0	0	0	0	64.5	[1, 7]
Q_2/I/A/C	0	0	5E-06	-0.0012	0.1205	-4.7599	93.614	[41, 83]
Q_2/I/A/T	0	7E-07	-0.0001	0.0058	-0.113	0.8709	62.273	[7, 54.09]
Q_2/I/β/C	0	0	9E-08	-3E-05	0.0002	0.0246	41.062	[0, 100]
Q_2/I/β/T	0	0	-6E-07	0.0001	-0.0126	0.5887	23.694	[0, 100]

Where:

I: Impeller.

D: Body or diffuser.

A: Vaned zone.

NA: Vaneless zone.

T: Shroud.

C: Hub.

β: Beta angle.

Many measurements were taken in laboratories on the known efficiencies of electric pumps and on the efficiencies with the prototypes developed in accordance with the geometry of the elements adapted to the stated polynomial equation.

As an example, thirteen results for different flow rates (Q) have been attached. In these examples, in all the new electric pumps, the same kind of body-diffuser has been used, and a first kind of impeller for flow rates (Q_1) between 3250 and 5700 and a second kind of impeller for flow rates (Q_2) between 5000 and 7000 litres/minute.

Q_1	Old hydraulic efficiency	New hydraulic efficiency
3250	69.64	75.164
3750	75.2	80.33
4250	78.2	86
4500	79.15	87.2
4800	80.3	88.37
5100	80.4	87.5
5400	79.9	85
5700	77.5	82.4

Q_2	Old hydraulic efficiency	New hydraulic efficiency
5000	75	85.43
5500	78	86.7
6000	79	86.2
6500	76.5	83.7
7000	72	79.366